TECHNICAL MEMORANDUMS

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

No. 776

A DISCUSSION OF THE SEVERAL TYPES OF TWO-STROKE-CYCLE ENGINES

By Herbert J. Venediger

Automobiltechnische Zeitschrift
Nos. 19 and 20, October 10, and 25, 1934

Washington
September 1935

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INTRODUCTION

The design of an engine hinges upon the use to which it is put. Thus, the more diversified the demands made on it, the more complicated its design and vice versa. For four-stroke-cycle engines, the structural design is comparatively clear and simple. Excepting special designs, the freedom of design is limited and the success of a design is more or less the result of careful planning and arrangement of the individual parts, such as valve gear, shape of combustion chamber, intake pipes, oiling system, etc. Even the arrival of the long- and much-discussed single-sleeve valve which is to supersede the orthodox poppet valve, would not enlarge the scope of design freedom, because it lies in the very nature of the four-stroke-cycle process that the development be limited or, to be more exact, be in a definite direction.

But for the two-stroke-cycle engine the conditions are substantially otherwise. Here the value of a design is judged in its totality rather than being primarily governed by the structural perfection of its component parts, which explains why it is of itself much more difficult to produce a serviceable high-powered two-stroke-cycle engine.

The three most important design factors are: volume of scavenge and charge delivery, scavenging process (scavenging result), and result of charge.

*"Planung und Aufbau schnellaufender Zweitaktmotoren." Automobiltechnische Zeitschrift, October 10, 1934, pp. 495-502; and October 25, 1934, pp. 529-535.

1. THE PRINCIPAL DESIGN FACTORS

The first of the cited factors, the "volume of scavenge and charge delivery," comprises, from the efficiency point of view, the scavenge input $\lambda_s = V_s/V_h$ (V_h = piston displacement, V_s = volume of scavenging medium inducted in cylinder), and the scavenge input power N_s (hp.) for induction, compression, expulsion, and transfer of scavenge medium as far as the exhaust passage. From the structural and operational point of view the factor includes: the selection, arrangement, and design of the scavenger, the design of inlet, transfer and scavenge ducts, the intake and exhaust ports or the diaphragms of the compressor, etc.

The second factor, the "scavenging process," concerns in particular the quantitative and qualitative scavenging efficiency η_s and η_s', that is, the ratio of inducted (V_s) to effective scavenging volume (V_n) remaining in the cylinder and the purity of the charge after accomplished scavenging η_s'. The operational part of this important factor includes all processes within the cylinder during scavenging; first of all the secondary problem of scavenging: the cooling of pistons, cylinders, spark plugs, etc., in engines using fuel-air mixture for scavenging the formation of condensation, the phenomena of backfiring in the scavenger, etc.

The third factor, the "result of charge," comprises "efficiency of charge" η_l' ($= V_n/V_h$) and "condition of charge"; that is, pressure, specific volume, temperature, compression ratio, quantity, and direction of motion (turbulence, etc.) of charge after consummated scavenging; the distribution of the fuel in the combustion air; and lastly, the important effect of the exhaust pipes on the process of charging and scavenging.

If it were possible to exactly combine all these design factors, their sum would give a criterion for the engineering value of the engine. Naturally, the number of feasible two-stroke-cycle engines is extraordinarily large because every one of the cited factors embodies a large number of design possibilities.

We begin with a brief discussion of the "scavenge result" factor. The multiplicity of scavenging methods suggested up to the present time is great while, on the other hand, a large number of these allegedly "original" scavenging processes may readily be traced to any one of the following well-known principles: transverse, reverse, counter, uniflow, and eddy flow process and their combinations.

The really practical scavenging methods, especially suitable for high-speed carburetor engines, are as few as the methods are numerous. This accounts for the widespread use of the so-called "transverse scavenging process" with specially designed pistons as late as 1932 in carburetor engines. Not until then did the two-stroke-cycle engine design (DKW) switch to the superior "reverse scavenging process" and specifically to the version where the scavenge ducts are substantially at both sides of the discharge passage and the scavenging medium is carried toward the rear cylinder wall (v. Schnürle's patents Nos. 511102, 520834), after the writer had pointed out in the early part of 1932 the serious defects of the conventional transverse scavenging method and the superiority and applicability, especially in the above version, of reverse scavenging in high-speed carburetor engines, particularly with crank-chamber pump (reference 1).

Unquestionably, there are other methods adequate for application to high-speed two-stroke-cycle engines, some of which are included in the report.

The third design factor, previously designated, for short, "result of charge", comprises all measures intended to prevent the loss of scavenging or charge volume and to assure the desired efficiency of charge $(V_n/V_h = \eta_l' \gtreqless 1)$, boost, or supercharge. In this category belongs the investigation of phase diagrams and devices in the scavenge, charge, boost, and exhaust pipes, the design of back pistons, U double piston, U-cylinder, and slide-valve engines; lastly, the investigation of the charging, boosting, and supercharging processes and the behavior of the charge up to firing. It is obvious that the structural possibilities for obtaining a predetermined phase diagram or certain charge efficiency, are extremely great.

The design possibilities afforded from the divers scavenging processes and the numerous charging methods are further enhanced through the possibilities hidden in the factor "volume of charge and scavenge delivery." This factor is of prime importance from the design point of view and implicitly governs the value, purpose, and construction, according as the requirements stipulate.

2. SCAVENGE AND CHARGE-DELIVERY VOLUME

The following arrangements come into question:

a) The design of the crank chamber of each working cylinder as crank-chamber scavenge pump (abbreviated KK).

b) Arrangement of auxiliary suction piston (for short, HK) linked to the working connecting rod, supporting with its lower side the suction effect of KK.

c) Arrangement of single (or double) acting piston-charge pump (KP) with exhausted crank chamber.

d) Arrangement of rotary scavenge and charge pump (RP) (rotary vane-type supercharger, Roots blower, etc.).

e) Combination a + c : KP + KK.

f) Combination a + d : RP + KK.

g) Design of working piston as differential or stepped piston (abbreviated SK) with crank-chamber pump.

h) Design of working piston as stepped piston, whose upper side operates when the crank chamber is exhausted (limited to multicylinder engines).

i) Combination g + h : arrangement of stepped piston with top and bottom suction (limited to multicylinder engines).

k) All combinations of a to i.

The combination of these chief possibilities in scavenging with the existing, particularly competent scavenge methods or charge, boost, and supercharge arrangements already affords several hundred kinds of two-stroke-cycle designs. Only a few of these actually "lead to Rome", and because this fact had been previously overlooked and is even today quite frequently disregarded, it has done great damage to the development of the two-stroke-cycle engine, has brought one especially useful principle into disrepute, and stamped it with the stamp of inferiority. The meritorious attempts of individuals - foremost among which is I. S. Rasmussen - form a glorious chapter in the history of engine research.

The investigation hereinafter treats the design possibilities from the concrete case of size and number of cylinders rather than from the cited viewpoints enumerated under a to k, although in the summary the other way is chosen. The combinations g to i were included for reasons of completeness, even though the stepped piston is, for the time being, without the scope of the problem, especially as concerns the high-speed engines (operating at 3,000 to 3,500 r.p.m.) under discussion.

3. SINGLE-CYLINDER ENGINES

a) Crank-Chamber Scavenge Pump

This very simple and inexpensive version of the two-stroke-cycle engine is manufactured by the thousands as carburetor and diesel engine. Scavenge input λ_s and volumetric efficiency of crank-chamber pump η_l are identical. Although the "clearance volume" of KK amounts to approximately 2.0 to 2.5 V_h, the volumetric efficiency is noteworthy and much higher than previously presumed or calculated. This fact is intimately connected with the vibration phenomena observed in the exhaust pipes, according to latest researches.* With correctly designed ports,

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*M. Kadenacy in Suresnes (Seine), on the basis of these occurrences, believes to be able to eliminate the KK altogether by fitting return flaps in the exhaust line (latest version of swirl-producing inserts) and to leave the induction of the fresh gas solely to the vibrating exhaust column. (Compare German patents Nos. 550283 (1927) and 550595 (1929). Naturally such work only at special r.p.m. (1,200 to 1,500) synchronized with the exhaust vibration, and are practically without significance despite the allegedly obtained b.m.e.p. of 6 to 7 kg/cm^2 and specific fuel consumption of from 180 to 200 g/hp./hr.

a maximum volumetric efficiency of from 0.60 to 0.70 is within the realm of possibility with the present-day r.p.m. and scavenging pressures of from 1.35 to 1.40 atm.

The vitally important quantitative scavenge efficiency η_s is greatly affected by the scavenging method. With transverse scavenging, it ranges from 0.70 to 0.75; with reverse scavenging and scavenge ports fitted at both sides of the discharge passage, it amounts to 0.80 to 0.85, that is, a charge efficiency of $\eta_l' = \lambda_s \eta_s = 0.45$ to 0.48 for the engine with transverse and $\eta_s = 0.52$ to 0.55 for the engine employing the reverse scavenge process. The latter thus promises approximately 10 percent higher efficiency, as borne out in practice; the specific fuel consumption itself drops in the same measure. The qualitative scavenge efficiency η_s' lies between 0.65 and 0.75 or, in other words, the charge is noticeably polluted.

Figure 1 illustrates the results with modern single-cylinder crank-chamber carburetor engines of from 100 to 350 cm^3 swept volume and reverse scavenging. The maximum mean effective pressures p_e are, as already pointed out, obtained with cylinders of approximately 200 cm^3 displacement, and the specific power output drops when this limit is appreciably deviated from either upward or downward. Today's maximum is around 4.1 kg/cm^2. The remarkable feature is its position at 3,000 r.p.m. (curves 175 and 200). According to this, the best liter performance today lies at 32 horsepower at 4,000 r.p.m., and the worst (100 cm^3) at around 21 horsepower. The best specific fuel consumption b_e is 380 g/hp./hr. (curve 200), and the poorest about 430 g/hp./hr. (300 cm^3 engine); both figures represent the optimum of the curves. But as already pointed out, these relatively high specific fuel consumptions of the torque stand, in themselves are misleading because the fuel consumption per mile is not inappreciably more beneficial, since the engine operates under more favorable conditions (throttle setting, lower scavenge pressure).

A comparison of previous publications (reference 2) on the performance data of high-speed two-stroke-cycle carburetor engines with transverse scavenging (see fig. 6, curve KK), reveals a rise of from 3.5 to 3.7 kg/cm^2 to around 4 kg/cm^2 in mean effective pressure, and a drop of from 500 to 400 g/hp./hr. in specific fuel consumption. Further improvements in mean pressures of 4.5 kg/cm^2 and consumptions of about 350 g/hp./hr. may be expected.

One important fact needs to be stressed, namely, that the crank-chamber engine affords greater freedom of choice of scavenge method than any other type. There is no difficulty in mounting the scavenge ducts either opposite or at either side or between or below the outflow passages. Adequate flow conditions can always be realized, which is not always the case with other scavenge systems as will be shown elsewhere in the report. Consequently, the crank-chamber engine is, contrary to widespread opinion, not an inferior but rather an extraordinarily organic solution offering great freedom, which may even be called perfect if it succeeds in removing its chief drawbacks: defective control possibility and poor idling. And the solution of these problems, which at the same time would raise the torque in the lower r.p.m. range, constitutes today the real problem of the crank-chamber engine.

b) Auxiliary Suction Piston (HK) Linked to Working Connecting Rod

This widely used method is shown in figures 3 and 4. Figure 4 is the phase diagram of the system figure 3, the opposed auxiliary piston being mounted equiaxially. To assure complete exhaustion of KK, the cranks of both pistons are set for total scavenge crank angle $2\varphi_s$ (as a rule about 120°) rather than at 180°. The auxiliary piston crank leads by $2\varphi_s$, hence lags 180° plus φ_s. Since the auxiliary piston must be as light as possible, the free mass forces are in reality not balanced in any of the designed engines even though it is possible to do so.

In the system figure 2, the delivery volume increases with decreasing angle δ; the method (fig. 2) in which the axis of the auxiliary suction piston does not pass through the center of the crankshaft which makes the reduction in angle δ possible, is patented by Schütte-Lanz (1919) - German patent 336601x*. It is especially preferred in small diesel engines. Of course, the upper side of the auxiliary piston is equally usable and may be used as booster pump, for example - (Patent 468646, Schnürle, 1925). While an excessive scavenge input serves no useful purpose in diesel engines, it is even more useless in the carburetor engine which uses the air-fuel mixture for scavenging.

* The x denotes that the patent has expired.

Above a certain bore of the auxiliary piston, no further rise in power output may be expected, whereas the fuel consumption increases abnormally unless a so-called asymmetrical phase diagram is provided (3d design factor: result of charge), as illustrated in figure 15 for a so-called U-cylinder whose exhaust ports close more or less before the scavenging ports.

The connection between scavenge input and efficiency of charge η_s' on the one hand, and the scavenging efficiency η_s and η_l' on the other, is shown in figure 5 for engines with the usual symmetrical phase diagram. It refers to a reversal flow-type engine, with the usual 1.0 to 1.25 stroke ratios. The charging range to $\lambda_s = 0.70$ is covered by the usual crank-chamber engine (KK), while λ_s may vary between 0.5 and 0.7 depending on the engine structure. Above that range the KK engine with auxiliary piston (HK) and the two-stroke-cycle engine without crank chamber but with piston-charge pump (KP) comes into question. The range of the rotary-pump engine (RP) properly begins at least at $\lambda_s = 1.6$ because the size of the piston-charge pumps would be excessive. Figure 5 refers to the effective scavenge input λ_s* not to the theoretical λ_{s_0}, which is merely defined by the dimensions of piston-charge pumps and rotary pumps, and which multiplied by efficiency η_l, give the effective λ_s.

Figure 5 manifests only a minor improvement in the degree of purity of cylinder charge η_s' at $\lambda_s = 1.6$, where the residual gases still amount to around 7 percent parts by volume. For this reason an effective scavenge input above $\lambda_s = 1.2$ automatically eliminates the high-powered carburetor engine which rather needs residual gases to prevent detonating combustion. From here on the shape of the curves in figure 5 concerns only high-speed diesel (and carburetor racing) engines, especially those with pressure injection, in which the air input is largely governed by the fuel distribution as, for instance, the Junkers with $\lambda_{s_0} = 2.4$. The charge efficiency η_l' practically ceases to increase above $\lambda_s = 1.8$. Complete filling of stroke volume as well as perfect purity of the

*Generally referred to the usual operating r.p.m. of 3,000 in sport, and 4,000 in racing engines.

fresh charge, would necessitate infinitely great scavenge input. For this reason none of the scavenging methods insures a complete scavenging.

In sport or racing engines the theoretical displacement of the linked auxiliary piston is chosen at approximately 1.0 to 1.25; V_h of the working piston and the clearance volume of KK at approximately 4.0 V_h. With symmetrical phase diagram and competent scavenge method, λ_s then lies at 1.0 to 1.1 for the necessarily high ports, so that in the extreme case (engines prepared for racing), a b.m.e.p. of 7.7 kg/cm² (3,500 r.p.m.) and an output per liter of approximately 65 hp., is obtainable (fig. 6). For comparison with this racing engine, we included the data for a slightly older KK engine. Both employ transverse scavenging with deflector. Naturally, the fuel consumption of the KK engine is much higher and amounts, in the particular case, to about 35 liters/100 km/1 liter stroke volume.

To assure high suction efficiency the stroke of the auxiliary piston is made considerably less than the bore; for example, $D_H = 2S_H$, $S_H = 0.6$ to 0.7 s (where s = stroke of working piston). The very high fuel consumption of such high-powered engines is in part necessary in order to keep the extraordinary heat development, which assures reliability, within bounds. With asymmetrical phase diagram (U-cylinder, for instance), the charge efficiency for equal scavenge input is not inappreciably higher, so that here an output per liter of from 85 to 100 hp. up to 5,000 r.p.m. may be obtained with careful design of all details.

Other than for racing, for which figures 3 and 15 have proved excellent (DKW), two-stroke-cycle engines with auxiliary piston are chiefly employed for sport vehicles. An output per liter of 40 hp. is readily attainable with $\lambda_s = \sim 0.9$. It also affords the advantage of utilizing the favorable high r.p.m. for the purpose of avoiding high scavenge losses, since the free mass forces can be balanced. Admittedly, the type shown in figure 15 is fairly expensive.

c) Piston-Charge Pump (KP) Arrangement

If the crank chamber is to be free from pressure - say, in order to be able to operate without mixture lubrication - the system of a piston-charge pump arranged equiaxially, rectangularly, or obliquely to the working piston is pertinent (figs. 7 and 8). Figures 9 and 10 show the most appropriate phase diagram for it. Arranged axially parallel, the pump crank leads at least through angle $2\varphi_s$, and lags for the same amount in the rectangular arrangement. These arrangements constitute, in point of fact, the prototype of the two-stroke-cycle engine and may be traced in principle as far back as 1838, where William Barnett in British patent No. 7615, proposed to precompress gas and air in separate cylinders and then to ignite the mixture in a common cylinder after slight compression by means of an incondescent platinum sponge.

Figures 7 and 8, while without the scope of single-cylinder carburetor engines, are so extensively employed in single-cylinder compression-ignition engines because KK is here eliminated and the types shown in figures 7 and 8, despite various advantages, are not substantially more expensive than those illustrated in figures 2 and 3.

The type, figure 8, appears to have been first suggested in Dobbins' U.S. patent No. 895928x* (1908); the crank of the charging pump is already designed as eccentric. Burtnett's U.S. patent No. 1475426 (1923), is a version with U-cylinders and obliquely fitted charge pump, while the rectangular arrangement and U-cylinder are cited in Chenard and Walker's French patent No. 611482 (1926) and Müller's Czechoslovakian patent No. 27428x* (1926). Finally, a special form of linkage of pump piston is given in Deutz's Austrian patent No. 117720 (1927). Even with the charging pump the advantages, at least with mixture distribution, are only fully realizable with asymmetrical phase diagram, as exemplified in figure 21.

The permissible theoretical scavenge input λ_{s_0} for carburetor engines with symmetrical phase diagram in normal operation, lies between 1.0 and 1.20. These figures give, at 3,000 r.p.m. and for a suction efficiency of charge pump of approximately 0.65, an effective λ_{s_0} of from 0.65 to 0.80, and accordingly in figure 5, a charge efficiency of approximately 0.60. About 25 percent of the

*The x denotes that the patent has expired.

scavenging medium is lost, so that theoretically, the specific fuel consumption of such engines must be correspondingly higher than that of four-stroke-cycle engines. With corresponding unsymmetrical phase diagram (for example, exhaust opens 25° before and closes 20 to 25° before inlet), λ_{s_0} may assume values as high as 1.5, as proved in figure 25. In order to obtain a high-suction efficiency, one prefers a 2:1 to 1.5:1 bore/stroke ratio of the charging piston, and at the same time employs membrane valves as in figure 8, or membrane valves and suction ports, in conjunction.

Junkers' engine (fig. 11) represents a special version of the equiaxial piston-charge pump. The four principal factors: unsymmetrical phase diagram, excellent scavenging, minimum bulk of scavenger, and free mass balance are very ingeniously combined. Unfortunately, this type does not lend itself to speeds above 2,000 r.p.m. in the arrangement shown in figure 11, because of the extremely oscillating and vibrating masses, except for the version of the still more expensive twin-shaft aircraft engine.

d) Rotary Scavenge Pump (RP) Arrangement

Its principle is illustrated in figure 12. It corresponds to Poyet's German patent No. 276783x* (1913), and to the still older patent of Zoller, No. 258173x* (1910), as well as to all of his expired patents of this kind which, in the majority, call for the impossible arrangement of a flywheel rotary vane-type supercharger, as in the German patent No. 363856x* (1918), for instance.

Since the delivery of the rotary compressor is continuous while the scavenging duct is open only for a third of the rotation, the arrangement concerns only multicylinder engines (at least two cylinders), or else undesirable pressure-balance tanks must be provided. For this reason the deisgn - in itself, original - of the flywheel disks, as rotor of the rotary vane supercharger whose casing forms its crankcase (see Zoller's patent No. 495997x* (1923)), is of no practical value. As concerns the volumetric, mechanical, and other characteristics of high-speed rotary vane-type displacement blowers, the reader is referred to a previous report (reference 3).

* The x denotes that the patent has expired.

e) and f) KP + KK and RP + KK Combinations

These combinations have not been tried as yet. The first does not even seem to have been proposed in the literature, while the latter combination - rotary vane-type supercharger with crank-chamber pump - is found in Zoller's patent No. 504909[x*] (1923). For apparent reasons these combinations have no practical significance. These designs obviously stipulate only symmetrical phase diagrams in diesel engines.

g) Stepped Pistons with Crank-Chamber Pump (SK)

Offsetting the piston, as in figure 13, ameliorates the volumetric efficiency of the crank-chamber pump because the clearance volume is substantially defined by the piston stroke. But, since the stepped part of the piston with its greater bore needs to be very long in order that its upper edge may overlap the inlet passage at bottom center, the piston becomes very heavy and the over-all height of the engine, excessive. To utilize the upper side of the step, it even has been suggested to allow air to be inducted through it which, at the exact moment, would be transported to the scavenging port in order to commence scavenging with air instead of with the fuel-air mixture (reference 1).

h) Stepped Piston Inducting Air at Upper Side

This system is applicable only to multicylinder engines since, owing to the phase lag between the exhaust period of the stepped cylinder and the scavenge period of the working cylinder, one cylinder must charge the other as illustrated in figure 26. As concerns the bottom side of the stepped piston, the conditions are the same as discussed under g) for the upper side.

i) Double-Acting Stepped Piston

Here the arguments adduced under h) are applicable. The bottom side of the piston would charge its own; the top, the adjacent cylinder. Even a minor enlargement as-

*The x denotes that the patent has expired.

sures a high scavenge input, so that this type is particularly suited for air compressing, even high-speed engines.

A novel type developed by the writer, suitable for high r.p.m., is to be discussed at the correct time.

4. TWO-CYLINDER ENGINES

a) Here the KK-type is also most prominent, as exemplified in the simple, inexpensive, and reliable DKW front-drive automobile.

The drawback of the usual KK engine is that the mass forces are balanced only to the first order, and the appearance of a free upsetting moment governed by forces of first order together with - as a rule - relatively low critical r.p.m. This makes it difficult to find a satisfactory suspension. Lastly, it has a vitiating no-load speed. All other types with 180° crank setting allow a very simple design and installation of additional charging devices.

b) Thus the KK engine with auxiliary piston needs only one auxiliary intake piston because, according to Schnürle's patent No. 471079x* (1924), its upper and lower sides aid in the suction and the supercharge (fig. 14). The phase diagram (fig. 4) remains unchanged. With adequately designed auxiliary piston, the power output which may be obtained is significant, according to the KK-HK curves in figure 6. They may even be raised considerably higher when an unsymmetrical phase diagram is chosen. (See fig. 15.) Figure 14 corresponds to the old, figure 15 to the modern, DKW racing engine.

c) KP engines afford a short but not simple design when both sides of the charge pump are utilized. The pump may be mounted on the end (fig. 16) or in the center (fig. 17). The former system engenders unevenly long scavenge ducts with their vitiating consequences (in volumetric efficiency, cylinder charge), which must be neutralized through different pump-chamber clearances. A further drawback is the uneven slope of the pipes - one upward, the other downward; this may easily produce different inlet conditions for the scavenge medium in the working cylinders. It may even make the use of many scavenge methods problematical, as detailed in section 6.

*The x denotes that the patent has expired.

Fitting the working cylinders on either side of the charge pump assures scavenge ducts of approximately even length but it does not remove the height discrepancies nor the uneven inlet conditions. Aside from that, the manufacturing difficulties in the construction and installation are also considerable.

All these drawbacks, and in addition the free upsetting moment of the types with double-acting charge pump, are voided in the design of figure 18 which (with regard to the phase diagram, fig. 9), gives a fairly exact picture of a fully symmetrical four-stroke 4-cylinder crankshaft. With adequate dimensions of piston and piston stroke, the mass moments can almost be eliminated while at the same time assuring perfectly equal charge and flow conditions in each charging and working cylinder. It represents the best two-stroke-cycle 2-cylinder engine with charge pump. It corresponds to Ringwald's patent No. 485889[x*] (1924). Superior, from the point of view of charge, to the type illustrated in figure 18 with the scavenge ports on either side of the discharge passage, is the type with reverse scavenging - the scavenge ports fitting between the two discharge passages - that is, facing the pump cylinders, as it affords the minimum possible pump-space clearance, exactly as for the engine employing transverse scavenging.

With charging cranks designed as eccentric, the rectangular or oblique charge-pump arrangement affords a very compact design, as illustrated in figures 19 and 20 where, for the sake of clearness, we show cranks instead of eccentric disks. Mounting two such single-cylinder assemblies side by side would afford an engine with high free forces and upsetting moments. This defect is surprisingly removed when the charging cylinder is mounted so as to be separate from its corresponding working cylinder by the other working cylinders, and the charge-pump delivery is crosswise as indicated in figure 20. This method, patented by Ringwald (No. 467676 (1924)) for diesel engines, has been known for some time on carburetor engines as, for example, through U.S. patent No. 1623391 of Burtnett (1925), and the Appleton engine of 1918, in which the charging cylinders were set obliquely.

In the arrangement figure 19 the scavenge-pressure lines may be freely shifted so as to assure high scavenging even without specially designed piston crown.

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*The x denotes that the patent has expired.

As the mounting of a charging pump in mixture compressing engines with symmetrical phase diagram, is a problematical matter even with the best scavenge method - at least with the scavenge inputs justifying a special charging device - it is no wonder that - apparently lacking other reliable designs - most patents refer to the combination of U-cylinder and charge pump as illustrated in figure 21. But the mass forces of such a high-speed engine are so great, even if in the form shown in figure 20, as to make smooth running practically impossible.

It suggests the design of the so-called "end-to-end" or "boxer" engine, the most simple and inexpensive version of which corresponds to the German patent No. 485889x* shown in figure 22 but which is inferior to the in-line engine (fig. 18), because of its high free forces and upsetting moments. These drawbacks do not exist in the design (fig. 23) where the mass forces are practically eliminated and the longitudinal upsetting moments are reduced to tolerable amount. Of course, such an engine is considerably heavier and more expensive than that shown in figure 22, where it is equally possible to use U-cylinders as shown in figure 24, in cross section through a separately acting charge and working cylinders.

Figure 25 shows the b.m.e.p. (p_e) liter performance N_L, and specific fuel consumption b_e obtained with 2-cylinder KK engines and two U-cylinder type engines with piston-charge pump and rotary pump (rotary vane-type compressor) corresponding to figures 20 and 21.

The KK engine gives, as in figure 1, a maximum b.m.e.p. of approximately 4.0 kg/cm², the U-cylinder engine with KP, a p_e = 5.3 kg/cm², while the U-engine with rotary vane-type compressor (U-RP) gives p_e = 5.75 kg/cm². In both U-type engines the theoretical scavenge input amounted to λ_{so} = 1.5. Owing to the strongly increasing volumetric efficiency of the rotary vane-type compressor beginning at 1,500 r.p.m., the torque of the U-RP type engine remains almost constant between 1,500 and 3,500 r.p.m., in contrast to its normal behavior in the KK and U-KP type engines. A glance at the p_e and b_e curves reveals the superiority of the U-KP over the U-RP engine; admittedly, above 3,000 r.p.m., the power output of the piston-pump engine increases very little.

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* The x denotes that the patent has expired.

No particular importance attaching to the combination d-f, nor to the Junkers engine in figure 11, which may be assembled in any number of cylinders, we proceed to combination g-i. The only version of this seems to be that shown in figure 26. Here one cylinder charges the other, although very unfavorably, because the cranks need to be set at 180°, which makes it impossible to realize the correct phase diagram (fig. 9). The pressure stroke of the charging piston has already ended when the scavenge ports of the other cylinder are fully opened. Consequently, on the down stroke the charging piston re-inducts part of the fresh charge. The stepped piston must be long enough to overlap the inlet passage E at top center; otherwise, it would fill the crank chamber. In the i combination, this is very necessary, as a result of which the stepped piston may be substantially shorter. Such an engine is therefore well suitable for higher r.p.m.

The design embodied in figure 26 is very old and well known as the Lutin or Coté engine, and Baer engine.

Another version is shown in figure 27. Here a U engine with cylinders showing the scavenge ports is fitted with a stepped piston whose upper side is inductive, so that there must be two mutually serving U-cylinders, as in figure 26, which are, in the known manner, placed obliquely to the axis of the crankshaft. Naturally, the new combination i is here also preferable.

5. THREE-CYLINDER ENGINES

Since odd numbers of cylinders preclude the use of double-acting auxiliary pistons and charging pumps, the 3-cylinder engine is restricted to comparatively few design types.

a) One simple and very serviceable type here is the in-line crank-chamber engine recently introduced in England by the Scott company. Since such engines manifest no free mass forces (balance up to fourth order) and the free upsetting moment with floating suspension (not necessarily Chrysler's "floating power") is not abnormally disturbed, aside from having an excellent torque and unlimited freedom of choice in the type of scavenge process as well as in the arrangement of the exhaust and scavenge ports, this type of design was to be primarily used in sizes of 0.7 to 1.2 liter swept volume for automobile and aircraft engines.

b) The 2-cylinder engine with auxiliary piston is practically ruled out because it requires three auxiliary pistons, and so does c) with its three single-acting charging-pump cylinders. Contrariwise, the arrangement d) of a rotary vane-type compressor is of interest because the charging periods of the three cylinders themselves are in consecutive order.

In contrast to the 2-cylinder, stepped piston-type engine of figure 26, the scavenging and charging in the 3-cylinder engine (fig. 28) takes place exactly in phase, thus realizing the optimum phase diagram according to figure 9, because the total scavenge angle $2\varphi_s$ is around 120°; that is, it agrees with the crank setting. The cylinders are mutually charged as shown in figure 28, with the noticeable drawback of one scavenge line being much longer than the other two. This renders a uniform charge of the cylinders difficult. In principle, of course, the statement made about the arrangement figure 26, applies here also. The arrangement figure 28, may be applied to a radial engine without the specific disadvantages, as seen from figure 29. This is the well-known Laviator design, of many years standing. An engine of this kind, built more than fifteen years ago, is said to have weighed only about 64 kg (141.096 lb.) with a 50 horsepower output.

6. FOUR-CYLINDER ENGINES

Starting with this number of cylinders the principal difficulties in design make themselves felt. Perhaps the chief advantage accruing from the use of few cylinders, i.e., rapid-firing sequence, leads - except for the crank-chamber rotary pump or Junkers system - to unnecessarily long, and only partially symmetrical crankshafts, difficult to balance as well as to complicated pressure lines, hence questionable scavenging, etc.

a) As to the KK engine, of the three possible partially symmetrical shafts and six firing sequences (figs. 30-32), only the crank design (fig. 30) is of interest because of its lowest mass moments from the rotating and oscillating masses; its firing order is 1423 or 1324. Admittedly, all three crankshaft designs are free from mass forces of the second order, but in spite of that, the rotation of shaft (fig. 30) is not as smooth as that of the four-throw shaft which, although afflicted with free mass forces of the second order, is nevertheless fully symmetrical and free from upsetting moments. The two-stroke-cycle in-line engine is therefore out of the question without floating suspension, at any rate less than the four-stroke-cycle engine. Each shaft design requires five carefully sealed-in main bearings which, in itself not abnormally difficult to accomplish, leads to an undesirable condition. These are the main reasons why no attempt has been made heretofore at manufacturing on a quantity basis in-line KK engines of liter/performance of around 25 hp. at 3,000 r.p.m.

b) The arrangement of no more than two double-acting auxiliary pistons involves - much more than with four auxiliary pistons (as, for example, Steiger-Gockerell patent No. 534252 (1930) proposed) - an enormous expense, even if used for special purposes (high-performance engine).

c) The single-acting charge pumps being, for ostensible reasons, out of the question*, the in-line type gives at least six crank throws, either in the sequence AAPPAA and PAAAAP or APAAPA (A = working cylinder, P = double-acting charging cylinder), as may be deduced from figures 16 and 17. Obviously, such crank shafts having poor balance and involving abnormal structural length, are to be rejected, with the result that the development leaned toward the V-type according to figures 16 and 17. (See also Paffrath's patent No. 515493 (1930) (expired).) A version of this type, in aggregations according to figure 16, is the high-speed two-stroke-cycle design of figures 33-35, which represents the well-known 1,000 cm^3 DKW engine of the "special" and "floating" types. Figure 35 is a cross section of the double-acting charging cylinder. The 90° setting of the four cylinders with pistons working from two throws set at 180°, which would insure adequate mass-moment balance, is disturbed by the throw of the charging pump so that the shaft is not altogether without some upsetting moment and must be balanced accordingly to a reasonable, mean value. Even so, the short shaft is remarkably free from critical r.p.m. Taken as a whole, it represents a very useful, compact, 4-cylinder design, with uniform firing sequence which, however, like all V engines,

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*See Berthaud's French patent No. 413126 (1910) (expired), with its 4-cylinder, in-line engine and two single-acting charge-pump cylinders arranged in V shape on both sides.

is expensive to manufacture and requires a number of special tools and equipment.

The use of U-cylinders (fig. 36), which really justify the expense of the charging cylinder since they assure high scavenge input, leads here to conditions which are not mechanically controllable. Whereas a complete moment balance is already difficult with ordinary working cylinders, it becomes practically impossible with the twin-piston arrangement. It is barely possible to balance the rotating masses, let alone the oscillating pistons acting on one single crank. Even the use of a rotary compressor which does not disturb the balance, leaves one mass moment rotating with crankshaft speed which shakes the engine back and forth. The result is, as with all V engines, compulsory floating suspension because, even with perfect moment balance, free, horizontal forces still remain to act in planes perpendicular to the shaft axis and tend to shift the engine back and forth in the four-stroke cycle of the revolution speed.

The results with an engine conformable to figures 33-35 (curves KP) and those with a 4-cylinder V type according to figure 36 with rotary vane-type compressor arranged front to front (curves U-RP) are compared in figure 37. The theoretical scavenge input of the KP engine is approximately $\lambda_{s_0} = 1.0$ (the effective λ_s at 3,000 r.p.m. is about 0.65) against $\lambda_{s_0} = 1.5$ and $\lambda_s = 1.0$ for the U-RP engine, corresponding to the propitious efficiency curve of the rotary vane-type compressor at high r.p.m. The p_e curves show that the mean effective pressure is almost the same as the λ_{s_0} values. The specific fuel consumption of the U-RP engine, although much better as a result of the unsymmetrical phase diagram, is far from satisfactory for the reason that such liter/performance must closely approach that of the 4-cylinder engine if the operation is to be economical. This, however, is not possible with the mixture-type engines in spite of the unsymmetrical phase diagram because the scavenging pressure is high and the input power of the rotary vane-type compressor, considerable. In racing engines this is of no significance, whereas the excessive heat loading is; our modern fuels are inadequate to cope with it. Thus the highly supercharged, reliable two-stroke-cycle engine still awaits considerable development.

Ostensibly the design of figures 16 and 17 can equally well be executed as W or X type, as illustrated in the German patent No. 515493 (expired). However, the result would be an engine with irregular firing order or else simultaneously firing cylinders. Consequently, the advantage of rapid-firing order and uniform torque of the two-stroke-cycle method turns out to be very disadvantageous if the engine is to be compact and simple: A blessing becomes a plague.

In the design figure 38 (6-cylinder engine according to fig. 17), the firing order is 12cab3; the setting is 45-90-45-45-90°; and the exhaust noise is accordingly irregular. Mounting the cylinder blocks vertically on each other, the firing order is 1(2c)a(b3)1; that is, regular below 90°, but cylinders 2 and c and b and 3 fire simultaneously. Again it becomes necessary to revert to other design types - this time to single-run or multiple-row radials, if compact multicylinders are desired. Before discussing these important types, we shall touch upon several pertinent factors closely connected with the arrangement of the scavenging ducts.

As already pointed out, one particular advantage of the KK engine is the absence of all difficulties as regards the use of different scavenge methods. The reason for this is, that the scavenge ducts run along the cylinder parallel to the cylinder axis, as a result of which they may be arbitrarily distributed over the cylinder periphery, fitted with appropriate manifolds for the ports and mounted according to wish and purpose. These possibilities are severely circumscribed when piston-charge pumps are used, because then the difficulties may make the choice of scavenge method rather limited.

Ostensibly the most promising design, both from the point of view of minimum pump-space clearance as well as scavenge-duct design - that is, with axially parallel, single-acting charge pump - is that shown in figures 7, 22, and 26, all of which employ the so-called "transverse-scavenging method." In figure 18, where the scavenge ducts are on both sides of the exhaust passages, these conditions are evidently less favorable. The two scavenge ducts, placed half-way around the cylinder periphery, must be carefully rounded off and smoothed, which means painstaking core making and expensive casting. For the double-acting charge pump (figs. 16 and 17), the difficulties are particularly great because here pitch, slope, and more or less successful rounding off follow each other in haphazard fashion which in carburetor engines, leads to the formation of condensation and r.p.m. dependence. Aside from that, the external smooth appearance of the cylinder block must be preserved.

Figures 39 to 43 show combinations of figures 16 and 17, with differently arranged passages. When transverse scavenging is resorted to, figures 39 and 40 afford comparatively simple passages. (See also dashed lines in fig. 33.) Figure 40 is superior to figure 39, with its passages of even length.

Figure 41 yields more difficult conditions which, however, may be ameliorated to some extent by artificially raising the pitch and slope of both lines predetermined in double-acting charge pumps in order to place the pipes parallel to the cylinder axis. In short, the beneficial aspects of the crank-chamber engine must be copied as closely as possible. These disadvantages do not exist in the transverse scavenge design of figure 42. The scavenge ducts lie between the two exhaust pipes. Aside from the fact of improved scavenging, the conditions are precisely as favorable as with figures 7, 22, and 26.

The disturbing factor of both exhaust pipes meeting at an obtuse angle, may be removed by resorting to unlike, yet relatively very similar transverse-scavenge methods in both working cylinders, as indicated in figure 43. The left cylinder is scavenged according to figure 42, whereas in the right cylinder, each one of the two scavenge flows describes a path on the order of a ram's-horn, somewhat as known from the Krupp patent No. 519427 (1923).

Mounting cylinder blocks in V shape according to figures 39-43, those of figures 39 and 40 are again the most pleasing and simple because figures 42 and 43 have four exhaust pipes, while figure 41 suffers from the previously discussed defects. Thus the transverse-scavenge method proves itself to be a positively worthwhile solution, even from the point of view of other than scavenge efficiency, and the investigation of the problems - merely touched upon herein - teaches that a scavenge method considered solely from the perspective of the processes occurring <u>within</u> the cylinder, is far from being an exhaustive treatment.

7. SINGLE- AND MULTIPLE-ROW RADIAL ENGINES

The design difficulties regarding firing order and mass balance discussed in the preceding section, are most easily removed when mounting the design types (figs. 7, 12, 16 to 18, 22, and 23) as radials, according to figure 44. This automatically precludes the use of the crank-chamber pump, and the scavenge and charge volume must be effected through single- or double-acting charge pumps mounted equiaxially with the working cylinders or else through one or more conveniently arranged rotary pumps at the front.

Such a 6-cylinder radial with nose turbocompressor was proposed by Zoller as far back as 1911 (reference 4). The unsuitable turboblower was later replaced by a rotary vane-type compressor.

A radial design built up according to figure 7 but employing U-cylinders, is described in Augustine's U.S. patent No. 1623296 (1927); and with working cylinders designed as U-cylinders, is found in Hirth's patent No. 386355 (1921) (expired). The advantages of the radial are evidently the regular firing order, adequate mass balance, even-length scavenge ducts fed from a central source, compact design length, and lastly, fairly reasonable manufacturing costs - for which reasons the two-stroke-cycle radial is of particular importance for aeronautical purposes.

To assure greater power units, the designs of figures 16 to 18 may be arranged as radial or, preferably, as X-type. Apart from the excellent mass balance there is, however, the drawback that cylinders of different rows fire simultaneously. For that reason, the second row of a two-row radial is set at 45° (fig. 45) with respect to the first row. Simultaneous firing in such two-row radials is only avoided when the number of cylinders is even, because with an odd number of cylinders, one cylinder of each front and rear row face one another. With X- and V-type four-row radials, the fully symmetrical four-throw four-stroke shaft is preferably replaced by the Cadillac crankshaft V 63 of the well-known 8-cylinder V engine (fig. 46), because this shaft design - of itself afflicted with the moments of the rotating and the moments of the second order in conjunction with the cylinder rows set at 90° and correspondingly designed additional masses (fig. 46) following the suppression of the moments of the first, second, and fourth orders, and the free forces of the first and second orders - appears superior to the fully symmetrical shaft with its known forces of the second order. The added advantage here lies in the fact that the simultaneous or multiple firing takes place in different cylinder rows.

Admittedly, such designs call for rotary pumps in front of or between the cylinders. The fact that a really practical blower of this kind is so far unavailable, constitutes the greatest drawback to all further development of high-speed, two-stroke-cycle engines, and it seems high time for private concerns, as well as the State, to supply the necessary means for developing blowers of high volumetric efficiency at low r.p.m., which are capable of producing adequate pressures up to 1 atmosphere, and are reliable even at 5,000 r.p.m., high isothermic and adiabatic efficiencies being assumed. The need for such a blower is just as pressing for the multicylinder engine.

8. MULTICYLINDER ENGINES

It was already pointed out in section 6 that the in-line design, with two single- or one double-acting piston-charge pump, according to figures 16-18, is practically out of the question for more than two working cylinders. The only possible multicylinder in-line type is that with rectangular or oblique arrangement of single-action, piston-charge pump, shown in figures 19 and 21 and described in Prini and Berthaud's French patent No. 413126 (expired). Even so, such an engine would be heavy and expensive - bulky, with faulty force and moment balance. Without serviceable rotary blower, the multicylinder in-line engine is positively inconceivable.

In one engine of this kind (see Zoller's French patent No. 595651 (expired) and his 1.5-liter 6-U-cylinder racing engine, so much discussed lately), the arrangement of the compressor or compressors in front has the drawback of unequal volume of charge in the individual cylinders. Since the scavenge ducts run straight past the cylinders, the charging mixture shoots partly past the just-opening cylinder, and that is so much more as the scavenge input is greater and the scavenge and charging pressure is higher. The cylinder fitted at the end of such a straight charging duct has, as known, the greatest charge volume, whereas the rest of the cylinders are more or less starved.

As a matter of fact, the design of the scavenge pressure pipes is, as stressed in section 6, nearly always the most difficult problem. For an in-line engine, such as Zoller's, it would therefore be better to pattern the design of compressors for charging every two or three working cylinders, according to figure 12. This would insure an exactly equal scavenge and charging process for the cylinders fitted to the corresponding compressor which, naturally, is so much more important as the cylinders are more supercharged; otherwise, insurmountable charge and heat difficulties would result. We shall not discuss other details at this time.

Other possibilities for high-speed, in-line engines are: five, six, or at the most, seven cylinders. All shafts being partially symmetrical (only the crank star is symmetrical), the balancing of the free forces is excellent for the odd-numbered crankshafts, although balancing of the mass moments from the rotating and oscillating masses is not afforded without auxiliary means as with the 6-cylinder crankshaft, in which the moments, even for two-stroke cycle, automatically disappear, at least with two preferred shaft designs.

The best 5-cylinder crankshaft is that shown in figure 47. The firing order is: 1 5 2 3 4 or 1 4 3 2 5. An equally good crankshaft is obtained when interchanging throws 4 and 5 and 2 and 3. The balancing of the free forces is given up to the eighth order.

The 6-cylinder crankshaft manifests two promising designs (figs. 48 and 49), in which the moments resulting from the rotating and oscillating masses of the first order disappear, while the free forces up to the fourth order are compensated. The firing orders are: 1 6 2 4 3 5 and 1 5 3 4 2 6 (fig. 48) and 1 6 3 2 5 4 and 1 4 5 2 3 6 (fig. 49).

The 7-cylinder crankshaft appears to be already afflicted with too many cylinders when compared with the satisfactory qualities of the 6-cylinder shaft - quite apart from the fact that the tendency to torsional vibrations increases, and partially symmetrical crankshafts are more expensive to manufacture. The compensation of the free forces has here already proceeded to the twelfth order, while the moments of the rotating and oscillating masses of the best crankshaft design (fig. 50) with firing order of 1 6 5 3 2 7 4 and 1 4 7 2 3 5 6 are about six times smaller than for the five-throw shaft of figure 49 (reference 5).

9. CONCLUSIONS

Owing to the great number of viewpoints governing the design of two-stroke-cycle engines, a comprehensive survey is practically impossible. For this reason, we shall attempt to correlate at least the principal factors.

To begin with, there is a deep underlying difference between the two-stroke-cycle and the four-stroke-cycle engines, due to the fact that the former - in contrast to the latter - is, and will remain for divers reasons, an engine with comparatively few cylinders. The chief reasons lie in the rapid-firing order and in difficulties encountered up to the present time with supplying the scavenge and charge volume. Somewhat exaggerated, the two-stroke-cycle engine is, thus, a type of design which by its very nature, is meant to be disposed with moderate mass balance. Moreover, the trends of development being rather indefinite, the designer faces new problems with every new design, for the solution of which any experience gained previously is of less use than when it pertains to the design of a four-stroke-cycle engine. The consequent result is a round-about development rather than one in a definite direction, as proved by the literature and the innumerable design propositions never actually tried out.

Separating the "wheat from the chaff" it may be stated that: The two-stroke-cycle engine with crank-chamber pump is today an economical and, for small, simple engines with one to three in-line cylinders and a displacement up to 350 cm^3 per cylinder, an excellently suitable design type. Apart from poor idling, whose amelioration constitutes its chief drawback, it affords everything within reasonable expectation. For cheap everyday motor transportation, this design - say, with three in-line cylinders and about 1-liter displacement - is the best suitable power plant.

In the version with opposed auxiliary piston (figs. 3, 14, and 15) with symmetrical or, better, unsymmetrical phase diagram, this design executed for two cylinders - up to 750 cm^3 - is excellent for high-powered engines, whereby theoretical scavenge inputs up to 1.5 for sport cars, and still higher figures for high-powered engines, come into account.

The single-acting, piston-charge pump for 1- or 2-cylinder engines (figs. 7, 18, and 23), concerns chiefly diesel or fuel-injection engines. The design (fig. 18) as 2- or 4-cylinder V engine, or even as multicylinder in-line engine, is superior to that of figure 16 or 17, for reasons of charging and better mass balance. Lastly, an ingenious solution is represented in figure 23, while right- or oblique-angle offsetting of charging cylinders is of primary interest with U-cylinders set athwart the crankshaft. But the fundamental value of the arrangements with single-acting, piston-charge pumps lies, as already stated, in the possibility of single-row and multiple-row radial designs - perhaps according to figure 45. The effective scavenge input for commercial engines should not exceed 0.7 with symmetrical phase diagram, and 1.0 for sport engines; with unsymmetrical phase diagram, it should not exceed 0.90 and 1.25, respectively. Supercharged engines, of course, may involve substantially higher figures.

The use of a rotary pump very nearly voids all design difficulties in comparatively simple fashion. In-line engines with very good mass balance, or short V- or X-types - single- and double-row radials, respectively, can then be designed; the first partly with several simultaneously firing cylinders, the latter with regular firing order. Besides, the charging and scavenging process involves no difficulties. The multicylinder, the best of which - the 6-cylinder in-line engine with rotary blower, especially in conjunction (as suggested in section 8) with several compressors as in figure 12 - is, next to the in-row radials (fig. 45 or 46), the best possible solution of a high-speed, two-stroke-cycle engine. But the in-line design has, so long as no asymmetrical phase diagram is obtainable otherwise, in more reliable manner, the added advantage of being usable in the U-cylinder version without difficulty. Junkers' excellent design, as well as its chief drawback, has already been discussed. A similar twin-shaft design with rotary blowers which assure very high r.p.m., is being brought out by the English firm, Jameson Engine Company (according to Allgemeine Automobilzeitung, No. 24, June 16, 1934).

The yet-to-be-developed blower must have the volumetric efficiency of a good, four-stroke-cycle suction stroke between 0 and 1,500 r.p.m., and above that, the efficiency of a satisfactory rotary vane-type supercharger. It must also allow for r.p.m. as high as 5,000, and the isothermal efficiencies should not drop more than 35 percent even at low back pressures and high r.p.m. The most economical operation with free exhaustion should lie around 800 r.p.m., and with rising back pressure the best economical operating point should so shift that the best operating point for each 0.1 kg/cm^2 pressure rise lies approximately 300 r.p.m. higher. Not one of the present-day compressors meets these requirements.

Finally, the question may be asked, what the future role of the high-speed, four-stroke-cycle engine will be with the further development of the two-stroke-cycle engine. At present, the four-stroke-cycle engine justly occupies the range beyond the one-liter swept volume. It may be assumed that it never will be completely superseded within the range between 1.2 and 2.5 liters, especially not in the form of the four-stroke-cycle design, which is scarcely more expensive than an equivalent two-stroke-cycle engine, but with better idling and greater economy, fitted with floating suspension which has come into use within the last few years. It may therefore be assumed that the high-speed, two-stroke-cycle engine will occupy the range of small and high performance, whereas the four-stroke-cycle engine will rule the range of the intermediate power outputs and requirements, which are characterized by smooth running.

With high-speed, two-stroke injection engines, the development of which should be pursued intelligently and consistently, the use of design a) - the single crank-chamber pump - is precluded because of the necessarily high air input. With the consistently increasing use of wood gas, which lends itself particularly well to the two-stroke cycle, the design types b) and e) to i) are inapplicable because of the fouling of the crank chamber. The development of a blower with the requisite qualities outlined in section 7, is therefore, from this point of view, also the most urgent factor in engine design.

Translation by J. Vanier,
National Advisory Committee
for Aeronautics.

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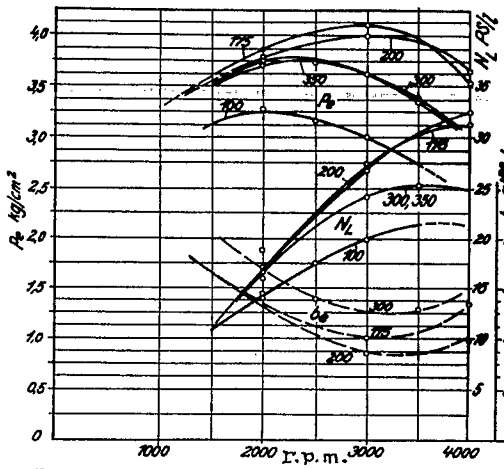
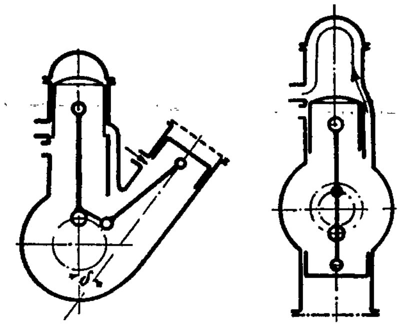


Figure 1.- Mean effective pressure p_e, volumetric performance N_L and specific fuel consumption b_e of modern two-stroke-cycle engines with crank chamber scavenge pump.



Figures 2,3.- Auxiliary suction piston arrangement.

Figure 4.- Phase diagram for fig.3.

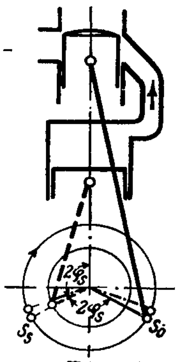


Fig.9

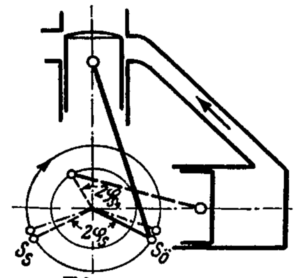


Fig.10

Figures 9,10.- Phase diagrams

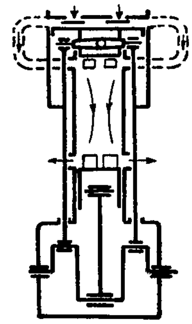


Figure 11.-Junkers type piston-charge pump.

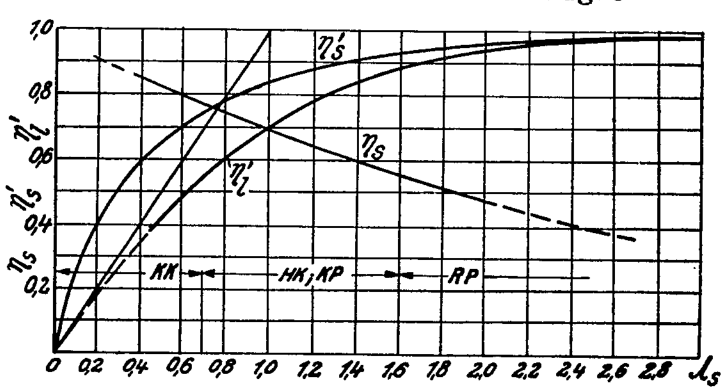
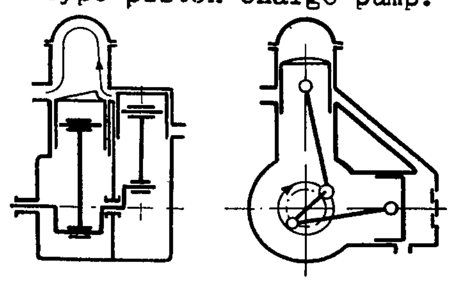


Figure 5.- Scavenge input λ_s, scavenge efficiency η_s, η'_s and charge efficiency η'_e with reverse scavenging.



Figures 7,8.- Piston charge pump.

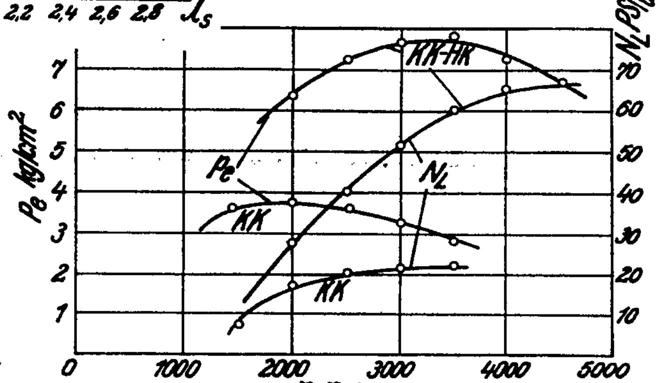


Figure 6.- p_e and N_L with crankchamber pump (KK) and auxiliary piston (HK).

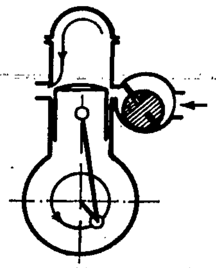


Figure 12.- Rotary scavenge pump principle.

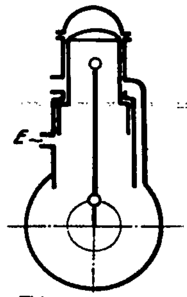


Figure 13.- Stepped piston.

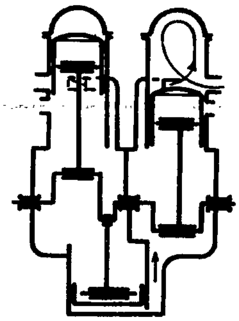


Figure 14

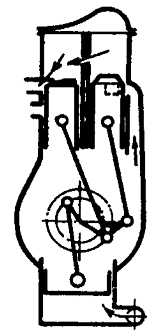


Figure 15

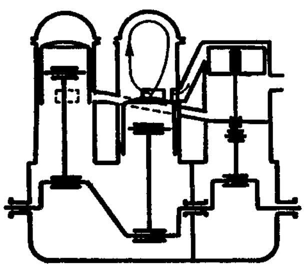


Figure 16

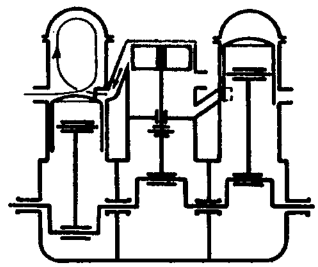


Figure 17

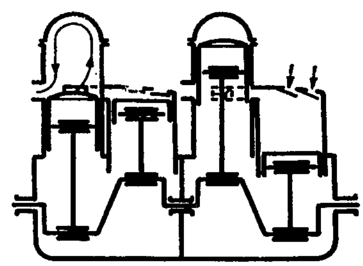


Figure 18

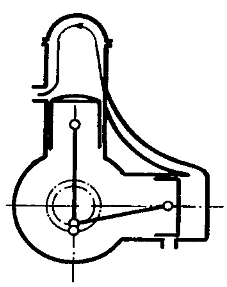


Figure 19

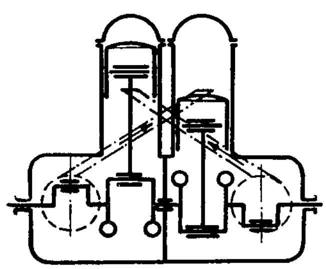


Figure 20

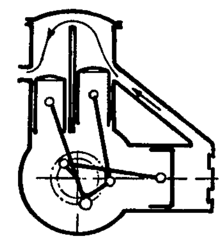
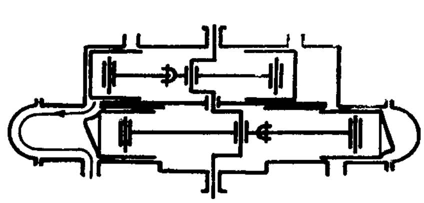


Figure 21



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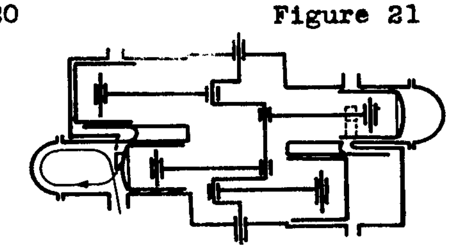

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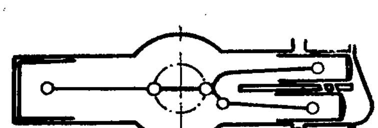

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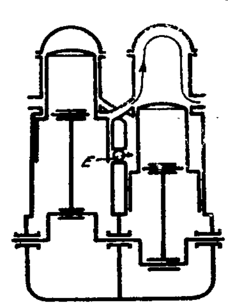

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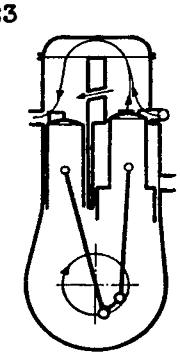

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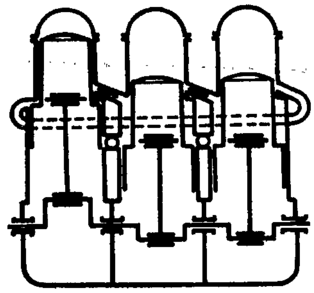

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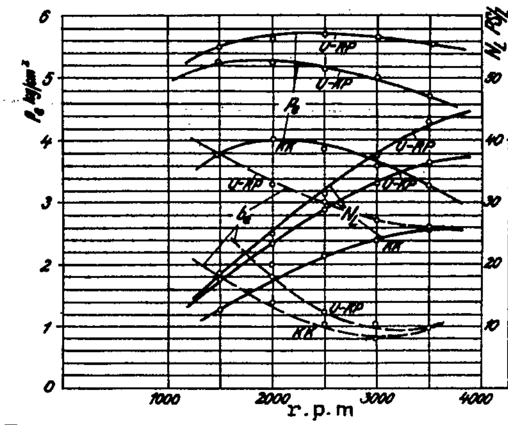


Figure 25.- p_e, N_L and b_e for multicylinder engines with crankchamber (KK), U-cylinder and piston-charge pump (U-KP) and U-cylinder with rotary vane type compressor (U-RP).

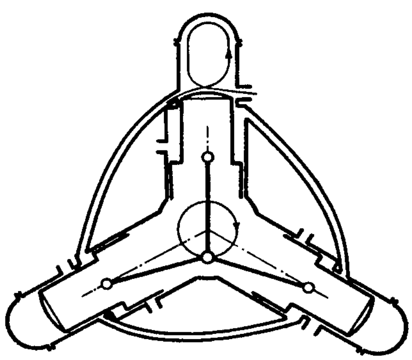


Figure 29

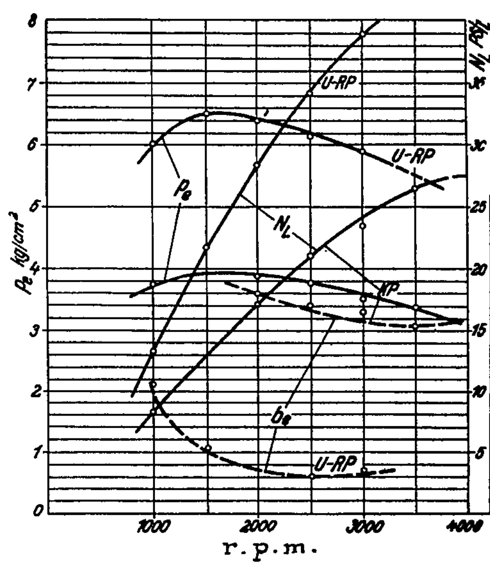


Figure 37.- p_e, N_L and b_e for four-cylinder engines with piston-charge pump (KP) and U-cylinder with rotary vane type compressor (U-RP).

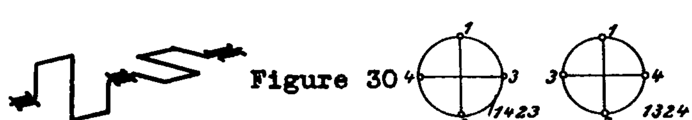
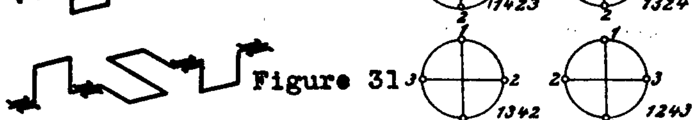


Figure 30



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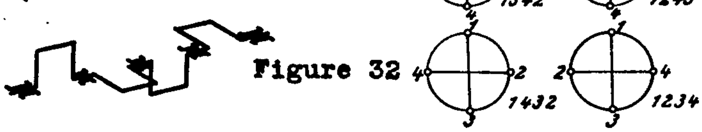
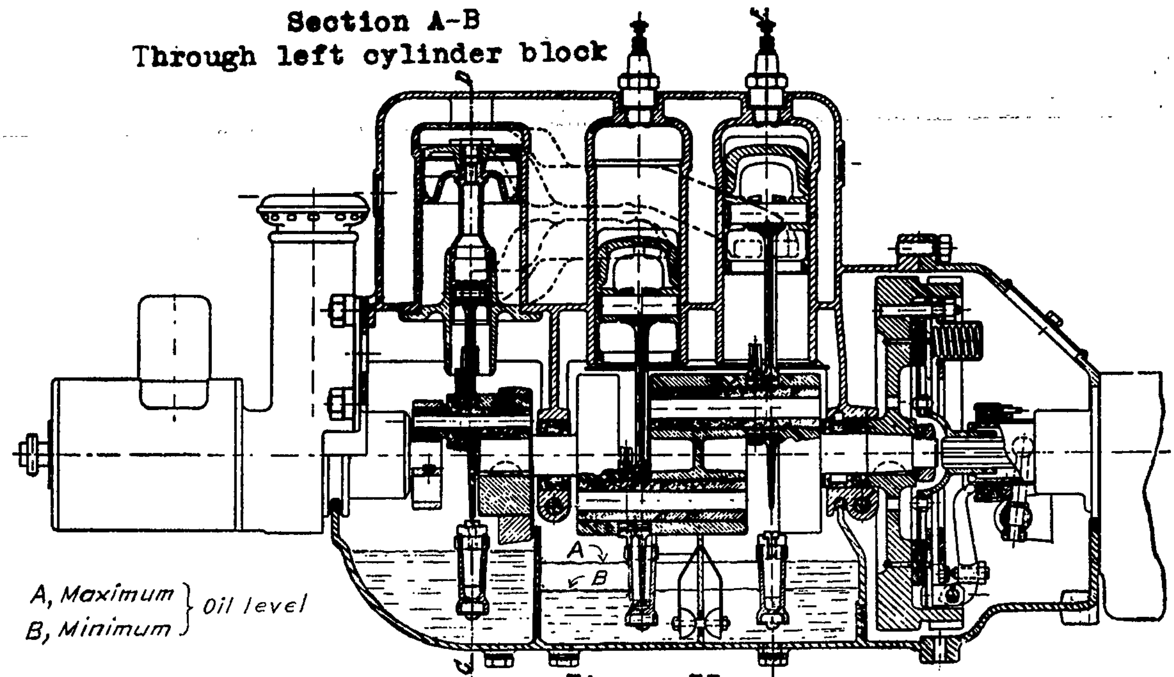


Figure 32



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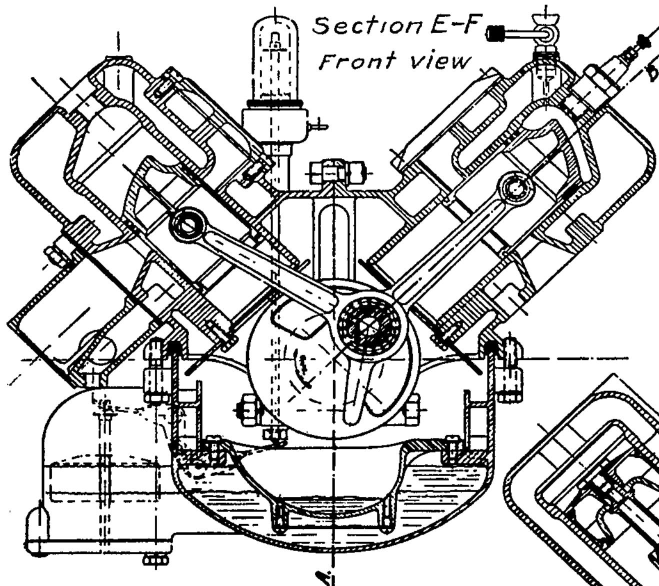

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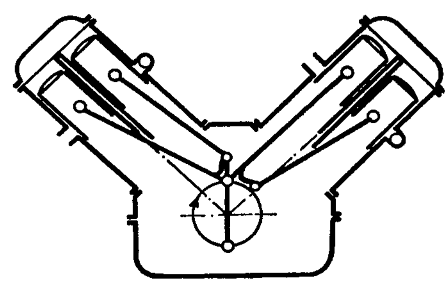

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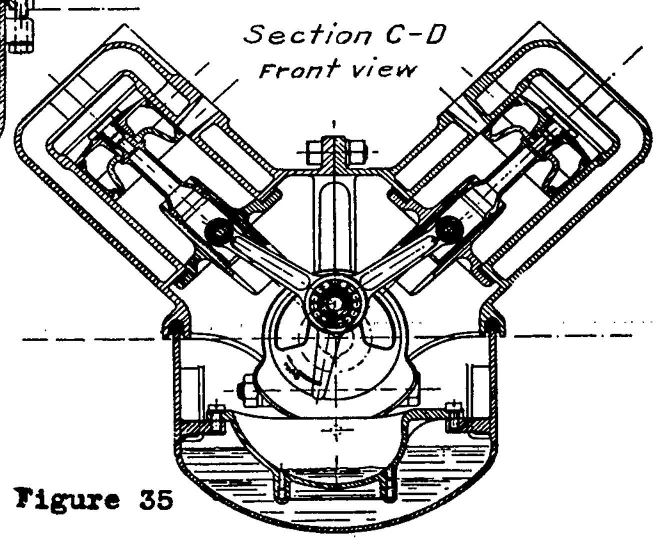

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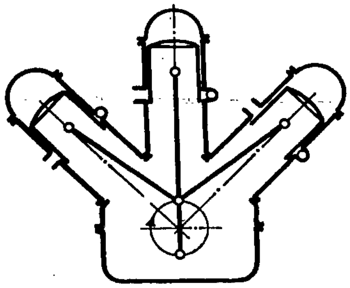

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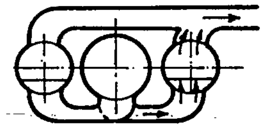

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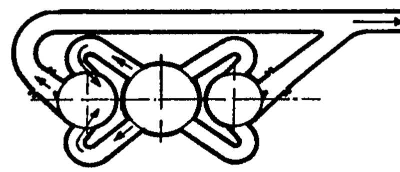

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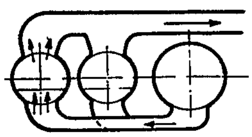
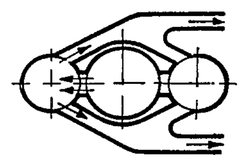


Figure 39



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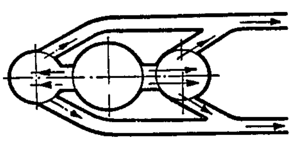

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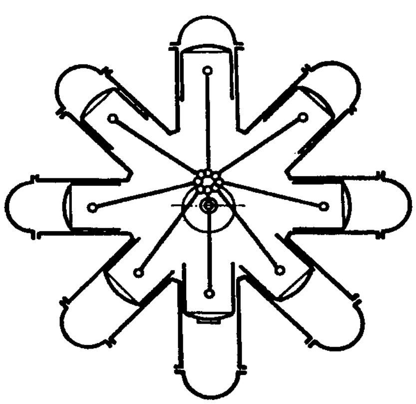

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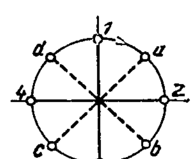

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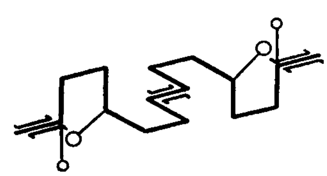

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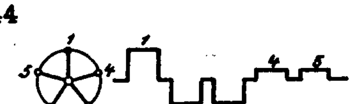

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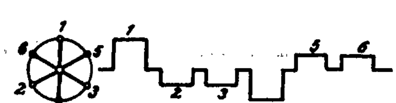

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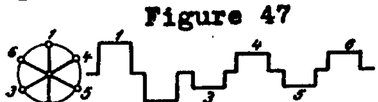

Figure 49

Figure 50